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DESCRIPTION AND PURPOSE

OF

TURBOMACHINERY FACILITY

by

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TABLE OF CONTENTS

HISTORY OF DEVELOPMENT OF TURBOMACHINES

The first practical steam turbines were built at the end of the last century. A Swedish engineer, C. G. P. de Laval, developed the first so-called impulse turbine which was tested in 1883. These turbines operated with supersonic flows for which he devised special nozzles which to this day are named after him. His turbines ran at speeds of up to 25,000 rpm with peripheral hweel speeds of over 1200 feet per second. De Laval recognized the phenomenon of critical shaft speed and used so-called flexible shafts which ran above the lowest critical speed. The first multistage reaction turbine was built in 1884 by C. A. Parsons in England. In 1898 the French firm of A. M. Rateau brought multistage impulse turbines on the market. It is of interest that the firms founded by these three engineers are still in existence.

In 1895 a British patent was issued to Parsons which clearly describes the operating principle of axial-flow compressors. A number of such machines were built by Parsons but their efficiency was only about 60%. Subsequently, centrifugal compressors were developed for the compression of gases which had efficiencies of about 70%. Their operating principle is similar to that of centrifugal pumps which was well known in the middle of the last century.

The first gas turbine was built in 1905 by the French engineer Armengaud. Its design was similar to present-day gas turbines but because of the low compressor and turbine efficiencies the unit was not able to produce any useful power. In 1909 the French engineer Marconnet received a patent for a propulsion unit for aircraft which is similar to a modern jet engine. Thus, around 1910 the operating principles of gas turbines and jet propulsion units were well known. However, the aerodynamic sciences had not reached a state

which made it possible to build turbomachines with high enough efficiencies which are necessary for successful power plants of this type. During and after World War I the knowledge in aerodynamics was increased greatly by theoretical and experimental work. In the period from 1920 to 1940 these advances were applied to the design of bladings of turbomachines, which in 1939 made possible the realization of the first gas turbine and the first flight with jet engines. Since then intensive research and development in turbomachines paved the way to supersonic flight and rocket propulsion. Although considerable advances have been made, new problems occur continuously. It is necessary to develop more efficient bladings both for compressors and turbines for jet engines of the supersonic planes of the future and for VTOL aircraft. The large liquid rocket boosters now under development require turbopumps of thousands of horsepowers which must be light and more efficient than conventional designs. Special pumps must be developed to produce the highest possible pressures at tremendous flow rates. Space power plants will have turbomachines with outputs of a few kilowatts to thousands of kilowatts, which must be light and have to run trouble-free for long periods of time at the highest possible efficiencies. Shaft seals and bearings must be developed that can operate under adverse conditions in space. Extreme wheel speeds are necessary to obtain high efficiencies, and the highest possible operating temperatures must be used to limit the size of the radiating surfaces that serve as heat sinks in outer space. Light and highly efficient gas turbines are required for helicopters, ground-effect machines, and hydrofoil vessels.

PURPOSE OF LABORATORY

Because of the very complex flow phenomena in rotating machines it is not possible to design turbomachines purely on the basis of theoretical

investigations. Theory shows the way for improvements, but for an actual design the engineer needs empirical data. Analysing the present knowledge of the losses and design parameters of turbines, it can be seen that there exist wide areas of ignorance even for present designs. Similar conditions exist in the field of compressors and pumps. Relatively small improvements in blading efficiencies produce large improvements in power plant efficiencies. In a gas turbine an increase of the compressor and turbine efficiency from 80% to 85% increases the output power of the plant by about 40%, at the same expenditure of fuel.

The heart of a turbomachine is its blading, and improvement can be reached only if the physical phenomena and the mechanism of the flow in such bladings are understood better. The turbomachinery facility of the Propulsion Laboratory has been laid out with this purpose in mind. Large scale cascades will be studied in the so-called Rectilinear Cascade Test Rig. The blades to be investigated are about 10 times the sizes of actual blades and they are arranged along a straight line instead of a circle, as in an actual axial turbomachine. These tests will permit a thorough investigation of the flow mechanism in such cascades, and establish criteria for the prediction of the flow losses and the separation parameters which in turn provide the required data for the best possible utilization of a blading. Because of the large size of the blades it is not possible to run these tests at the Mach numbers that occur in actual machines. An available motor and fan with a power of 700 HP limits the Mach number of the flow in the Rectilinear Cascade Test Rig to about 0.4. In this test rig it is further not possible to simulate exactly the flow conditions in an actual machine where the blades are installed radially on a cylindrical hub, and where stationary and rotating rows of blades are arranged adjacent to each other. In so

called Cylindrical Cascade Test Rigs large scale cylindrical cascades will be investigated. The test rig will have blades of the same profiles and geometry as those tested in the rectilinear cascade rig to establish correlation between the flow phenomena in rectilinear and cylindrical cascades. Theoretical considerations show that the flows in rotating and stationary cascades are fundamentally different, and that it is not possible to simulate the conditions that occur in rotating rows in a stationary cascade rig. For this reason it is planned to test stationary and rotating cylindrical cascades. The same fan which supplies air to the rectilinear cascade rig will be used for this purpose also.

Except for a few isolated cases, the design data for rotating cascades of turbomachines have been established by tests of stationary, and particularly by rectilinear cascades withou investigating whether these data are really applicable to the actual rotating rows of blades. The primary purpose of the cascade test rigs of the Turbomachinery Facility of the Propulsion Laboratories is therefore to provide fundamental information on cascade design parameters, and to establish correlation among the data obtained from rectilinear, cylindrical, and rotating cascades.

Since these cascade tests have to be undertaken at relatively low Mach numbers, it is necessary to test actual machines at high speeds and high Mach numbers, having bladings geometrically similar to those evaluated by the cascade tests. Such tests will be performed in the so-called High-Pressure Laboratory. A 1250 HP axial-flow compressor with a pressure ratio of 3 is used as the source of air to operate turbines, either directly for their tests or as a drive for compressor stages. Because of the large flow rate of the installed axial-flow compressor it will be possible to use only

part of its air supply for the drive of a test turbine, and to use the remainder of the flow rate as energy source in an ejector system which is capable of maintaining a vacuum at the turbine discharge. This ejector system will produce pressure ratios of about six, so that turbines stages with supersonic flows of Mach numbers of about 1.8 can be tested in this facility. Such test turbines will be used also as the high-speed drive for tests on compressor stages, and on bearing as well as on seal test rigs.

In its completed state, the Turbomachinery Facility will make it possible to undertake systematic investigations of bladings of turbomachines, beginning with the simplest tests with rectilinear cascades and followed by tests with cylindrical cascades, either stationary or rotating, which will then be complemented by tests on an actual high-speed machine whose bladings are geometrically similar to the previously tested cascades.

Although this procedure in going in steps from a simplified model to an actual design is in keeping with good engineering practice, there does not exist another facility where this logical approach is possible or where attempts are being made to carry it out. Some institutions furnish cascade data, others give performance values of turbomachines, but there do not exist published data which show the correlation between the results of cascade tests and machine tests. It is for this reason that the engineer lacks the fundamental data for the design of novel turbomachines. In most cases it is necessary to fall back on incomplete data obtained from previous executions, and it is necessary to use trial-and-error methods to further the state of the art.

A particular turbomachine must be designed for fixed operating conditions. However, since the machine is usually only a component of a complete power plant, it is invariably necessary to be able to predict the performance of the machine at off-design conditions because of changes that might or do occur in the other components. In this direction the present state of the art in turbomachines is extremely deficient. If, on the other hand, the off-design performance of cascades were known, the establishing of off-design operating conditions could be carried out with comparative ease.

The facility at the Propulsion Laboratory has been planned to provide the information that is lacking in the field of turbomachiner. It can become an important center of activity in this endeavor, capable of furthering the state of the art. Besides being of benefit to a particular segment of American Industry, such efforts could help maintain the superiority of the United States in the field of turbomachines, and thereby reduce development costs for projects of the United States Navy in particular, and the Nation in general.

Turbines at high temperatures are limited in speed by centrifugal and thermal stresses. Very little information is available about design criteria in this respect. For this reason, and to spin the rotors of machines to be tested in the cells, a hot-spin test rig has been installed which is designed to measure stresses and strains in rotors at high speeds at at imposed temperature gradients from the center to the outer periphery of a rotor.

DESCRIPTION OF LABORATORY

Figure 1 shows the outside of the Cascade Laboratory building. Available is a blower which is capable of delivering an air flow of about 100,000 cubic

feet per minute at a pressure difference of about 45 inches of water at 100% speed. Since the blower motor, with a nominal rating of 700 HP, can be run at 50% and 75% speed, and because it has controllable inlet guide vanes, it is possible to reduce the pressure differential of the blower to about 6 inches of water. Within this range the Reynolds number of the flow through the cascade rigs can be changed by a factor of 3.

Figures 2 and 3 show the Rectilinear Cascade Rig with the plexiglass windows for the observation of the flow. In Figure 4 one of the side walls is removed to show the general arrangement. In contrast to conventional test rigs, the flow entering from the plenum chamber through a rectangular cross section of 10 inches by 60 inches is deflected by a closely spaced cascade to produce the correct inflow direction ahead of the cascade to be tested. By means of adjustable and telescopic side walls it is possible to change the inflow direction between 15° and 135° with respect to the axis of the cascade. Flow deflections in the cascade to be measured can therefore vary between a few degrees, as in axial-flow compressor bladings, and about 150°, as in extreme cases of turbine bladings. While in conventional cascade test rigs it is necessary to remove the boundary layers at the walls to produce uniform inflow conditions, the present rig allows measurements both without and with boundary layer removal, since each fluid particle travels the same distance from the inlet guide vanes to the actual cascade. Fig. 4 shows slots which are used for traversing the probes for the measuring of the flow conditions ahead of and after the cascade. These probes are mounted on the traverses which can be seen in Figs. 5 and 6. By means of a servo-mechanism the probes are automatically rotated into the correct flow direction. Both the total and the static pressure are measured

by transducers. Potentionmeters measure the location of the probe in the flow channel, both in lengthwise direction and across the flow channel, and the flow angle. Traversing is effected by electrical contacts from the console of Fig. 7. An analog-to-digital converter displays the measured values on a light panel, prints them out, and punches a paper tape which is used to feed the measured data directly into the CDC-1604 computer of the Postgraduate School, wihtout the necessity of preparing punch cards. Manual input data can be introduced also to establish the control and definition of the computer program, which evaluates the performance parameters of the test cascade in a few minutes.

An important feature of the data logging system of Fig. 7 is the calibrating procedure. It is possible to interrupt the measurements at any time to correlate a known pressure with the read-out of the system. Not only is it possible to compensate for any drift of the electronic circuits but to have the printed data in actual engineering units, and not only as electric outputs of the transducers.

The High-Pressure Laboratory of Fig. 8 is a building with two explosion-proof test cells of 14 feet by 18 feet, an open cell with the same dimensions, and a sound-proof control room that serves all three cells. Adjacent to the cells is the compressor room of Fig. 9 with a 12-stage axial-flow compressor having a pressure ratio of three and a suction flow rate of 10,000 cubic feet per minute. Its nominal speed is 12,000 rpm, but a fluid coupling installed between the 1250 HP motor and the speed-up gear makes operation possible in the range from about 5000 to 12,000 rpm. After passing through a cooler the compressed air flows to the different test cells. Since it is desirable to operate the test specimens in the cells with arbitrary flow rates, an automatic surge suppressing device is installed which by-passes the necessary flow from the compressor discharge to the compressor inlet. Control of this

device, and the operation of the compressor for different speeds and pressure ratios, is effected by means of a special analog computer, which calculates the surge limit from the measured compressor speed, the suction temperature and the pressure ratio, and maintains the controls at the desired operating point.

As shown in Fig. 10 the axial-flow compressor is equipped with static pressure taps ahead of and after each row of blades; by means of Kiel probes it is possible to measure total pressures and temperatures after each rotor. Hence, interstage data can be taken to correlate theoretical off-design performance data with experiments.

A data logging system similar to that of Fig. 7 is installed in the control room. It is primarily used to measure a multitude of pressures by means of scanner valves. At present only four such devices, with 24 pressure lines each, are installed; but the console can ultimately be equipped with ten such valves. Each valve scans 24 pressures with intervals of ½ seconds, so that it is at present possible to measure and record 96 pressures within about one minute.

In the open test cell of the High-Pressure Laboratory a 3-stage low-speed axial flow compressor is being installed, where accurate measurements of the flow phenomena in compressor stages can be undertaken.

One of the explosion-proof cells is used for a transonic turbine test rig which has been designed at the USNAVPGSCOL with the support of BuWeps. In this single-stage test rig it is possible to measure directly the moment and the axial force that are exerted on the guide vanes. Fig. 11 shows a view of the rig from the top, with the flange through which air from the

compressor enters a stationary casing. In Fig. 12 is shown the guide vane assembly, which is supported on a cradle by means of flexures. Fig. 13 represents the assembly of the test rig without the top of the hood, and in Fig. 14 is shown the complete unit with the exhauster.

Without the exhauster system the turbine will operate at a pressure ratio of three, developing about 200 HP at 20,000 rpm. The power is absorbed by an air dynamometer with remote control and remote read-out. The moment and the axial force exerted on the guide vane assembly are measured with reluctance type force capsules that change the frequency of an FM system. Static calibrations showed that the moment measurements are not influenced by the magnitude of the axial force. From the moment and the axial force that act on the guide vane assembly, and the flow rate, the actual average velocity at the guide vane exit and its average direction can be determined. Measurements of the static pressure between guide vanes and rotor blades establish the theoretical velocity at the discharge. These quantities make it possible to determine the losses in the guide vanes, without the need of making surveys by means of probes.

With the exhauster system a pressure ratio of about six can be maintained across the turbine, and its power will be about 360 HP at 20,000 rpm. Two types of guide vanes are presently available, one which has converging flow passages with the minimum cross section at the trailing edge, the other with converging-diverging flow passages with a discharge area about 120% of the minimum flow area. To be investigated is the pressure ratio at which it is advisable to arrange converging-diverging nozzles; or where the losses due to after-expansions in a purely converging nozzle become larger than the

frictional losses in the longer converging-diverging nozzles. Of particular interest in this study will be the interdependence between operating efficiency and the axial and radial blade clearances. The axial clearances between guide vanes and rotor blades can be varied with ease, and the radial clearances at the rotor tips will be changed either by grinding off the rotor, or by means of inserts. Two kinds of rotor bladings will be investigated. One type of blade profile consists of circular arcs and straight lines, whereas the other has carefully chosen contours without sudden changes in curvature. It will also be investigated how much rounded leading edges of the rotor blades influence the efficiency and the off-design performance.

The characteristics of the non-diverging nozzles and the rotor blades will be established in the Rectilinear Cascade Test Rig at low Mach numbers. These model blades are available and will be tested in the near future, to establish correlation between cascade data and actual performance at high Mach numbers.

In the third test cell a radial dual-flow turbine is installed. The description of this equipment, and of the test data obtained thus far, are given on pp. 76 to 96 of Ref. 1, Part III.

A Hot-Spin Test Unit is located outside the High-Pressure Laboratory. A reinforced pit in which rotors with diameters up to 50 inches can be spun is evacuated to an absolute pressure of 50 microns of mercury. An air turbine with a maximum speed of 50,000 rpm is used to rotate the test specimen. By electric heating at the outer diameter of a disk, and by radiative cooling of its parts near the axis by means of a water circuit, it will be possible to obtain temperatures of about 1800°F at the rim and about 800°F near the axis, to simulate the actual operating conditions of a turbine disk. On top of the air turbine is arranged a slip-ring assembly which connects a 12-channel

strain recorder and a 12-channel temperature recorder to sensing elements on the rotating disks. The spin unit and its measuring equipment will be used to undertake fundamental research on rotating disks of unconventional design to establish design criteria, especially with regard to therma shocks and thermal cycling.

REFERENCES

1. M. H. Vavra, "AGARD-VKI Lecture Series on Problems of Fluid Mechanics in Radial Turbomachines," Course Notes 55a and 55b in four parts, von Karman Institute for Fluid Dynamics, Rhode-Saint-Genese, Belgium, March 1965.

Figure 1: Cascade Laboratory Building

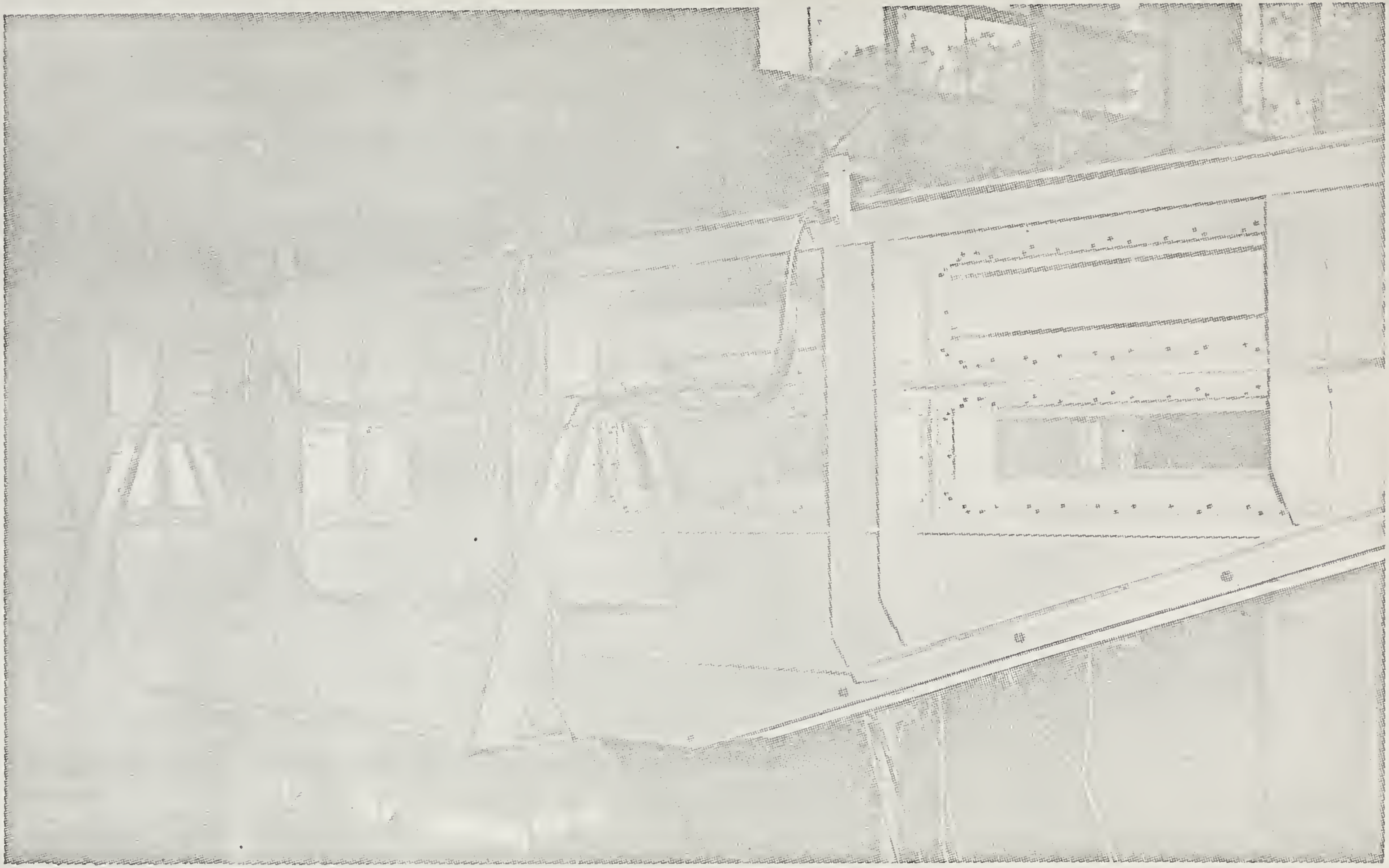

Figure 2: Rectilinear Cascade Test Rig

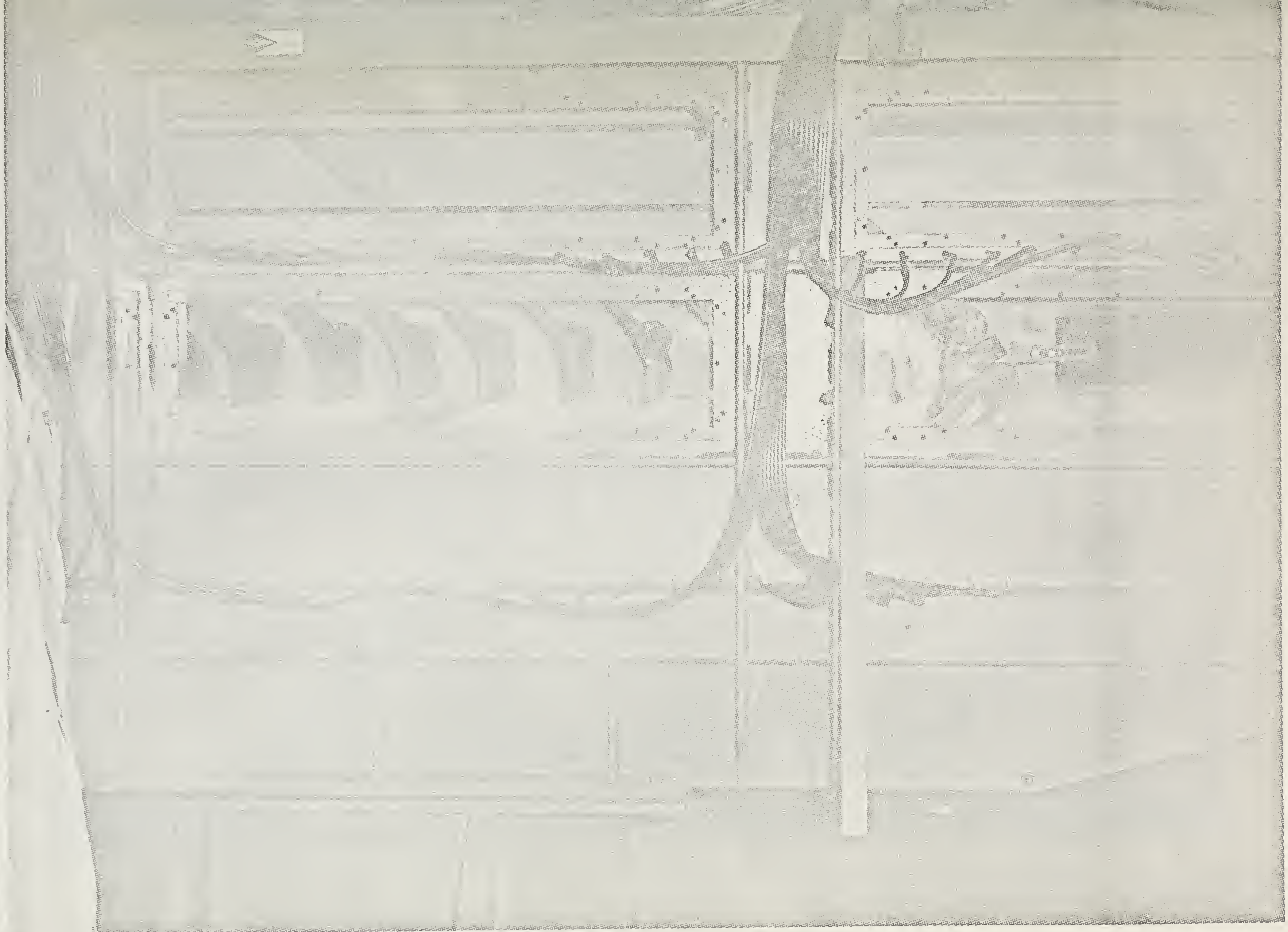

Figure 3: Turbine Cascade seen through Plexiglass Windows

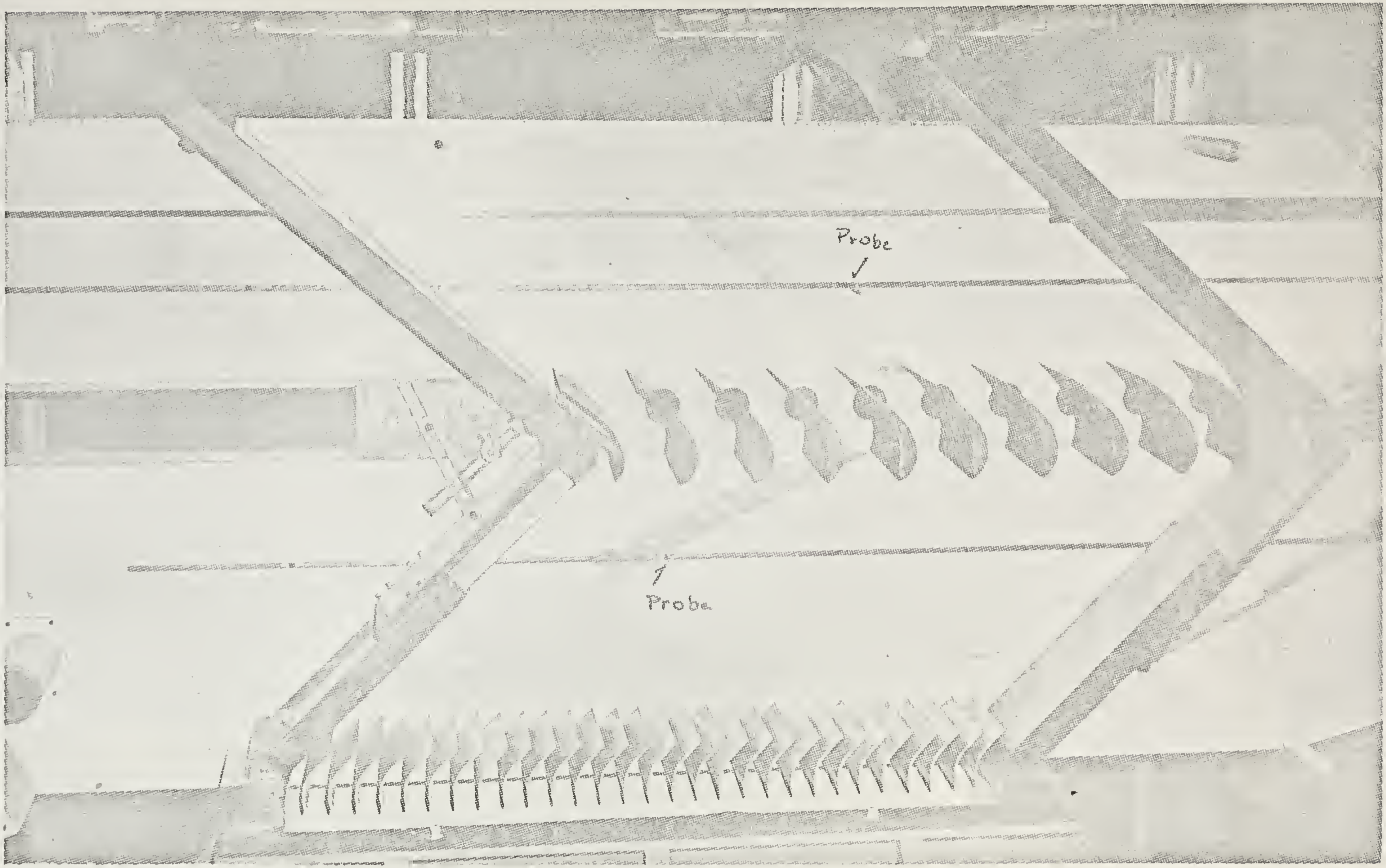

Figure 4: Arrangement of Turbine Test Cascade

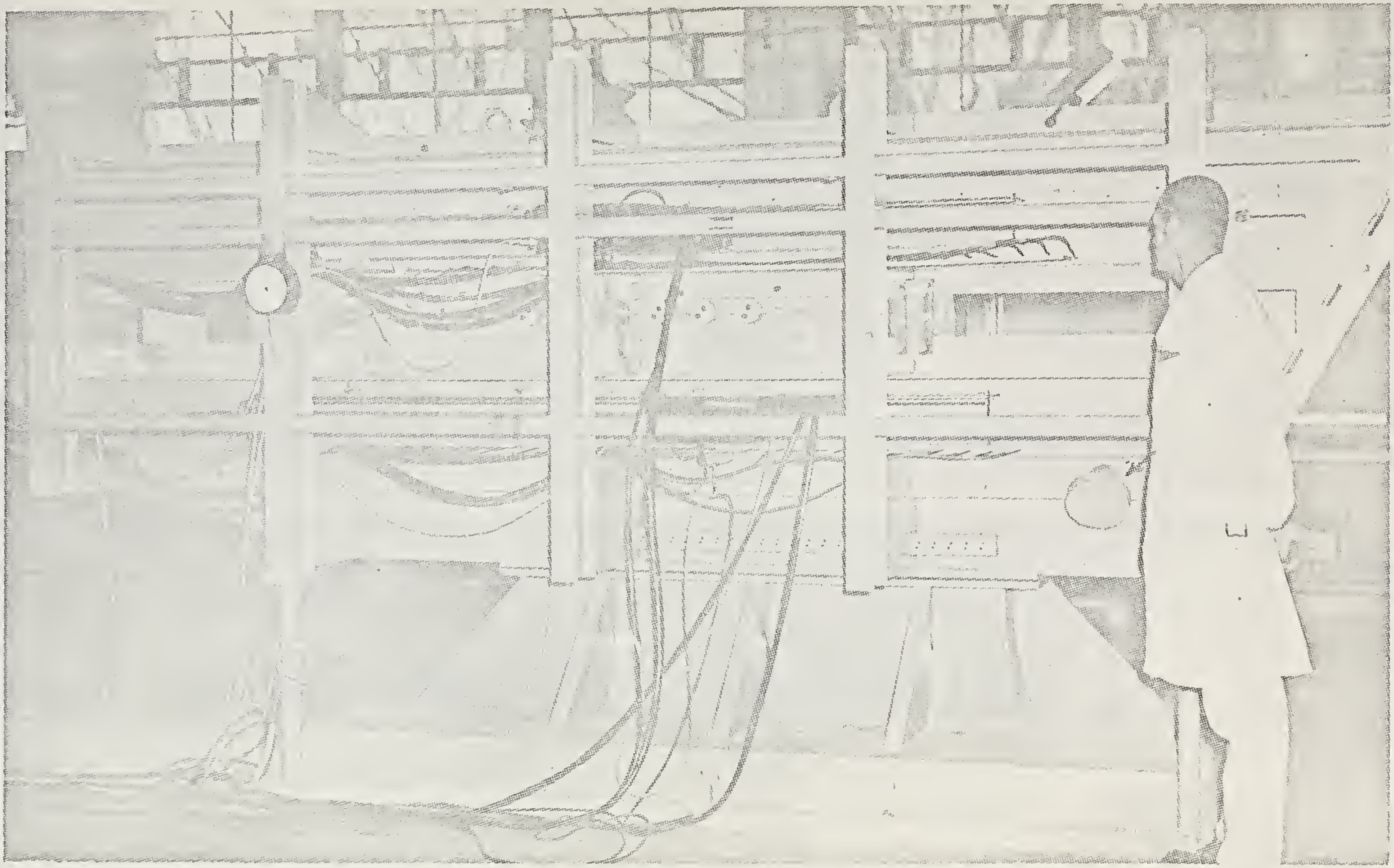

Figure 5: Rectilinear Cascade Test Rig. View of Traverses

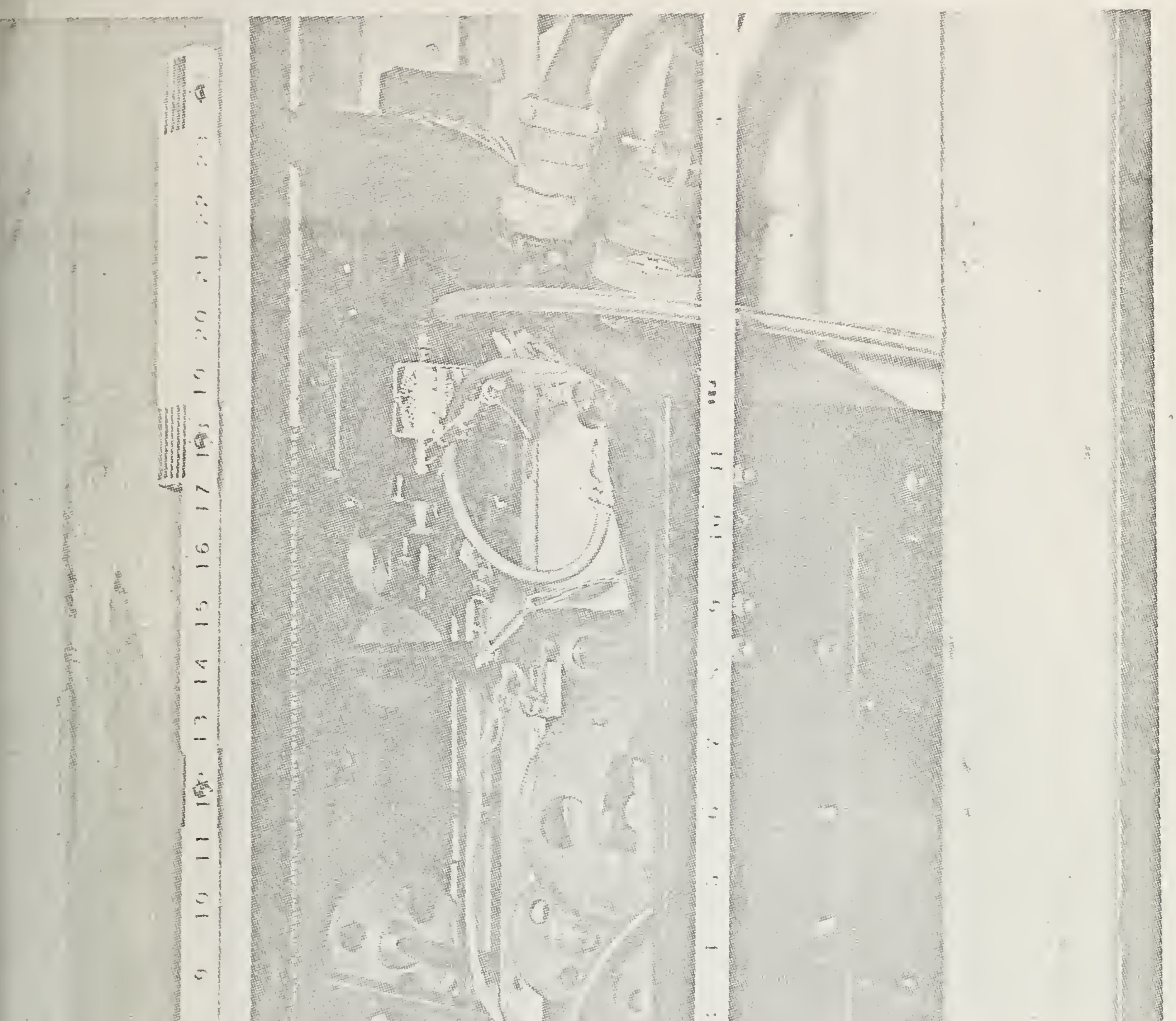

Figure 6: Measuring Traverse

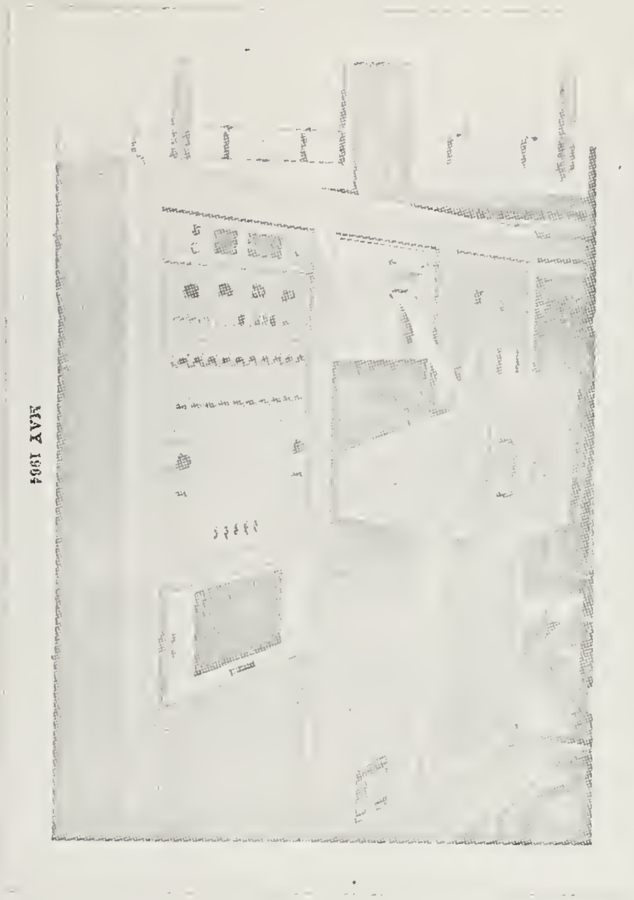

Figure 7: Automatic Logging System

Figure 8: High-Pressure Laboratory

Figure 9: Compressor Room

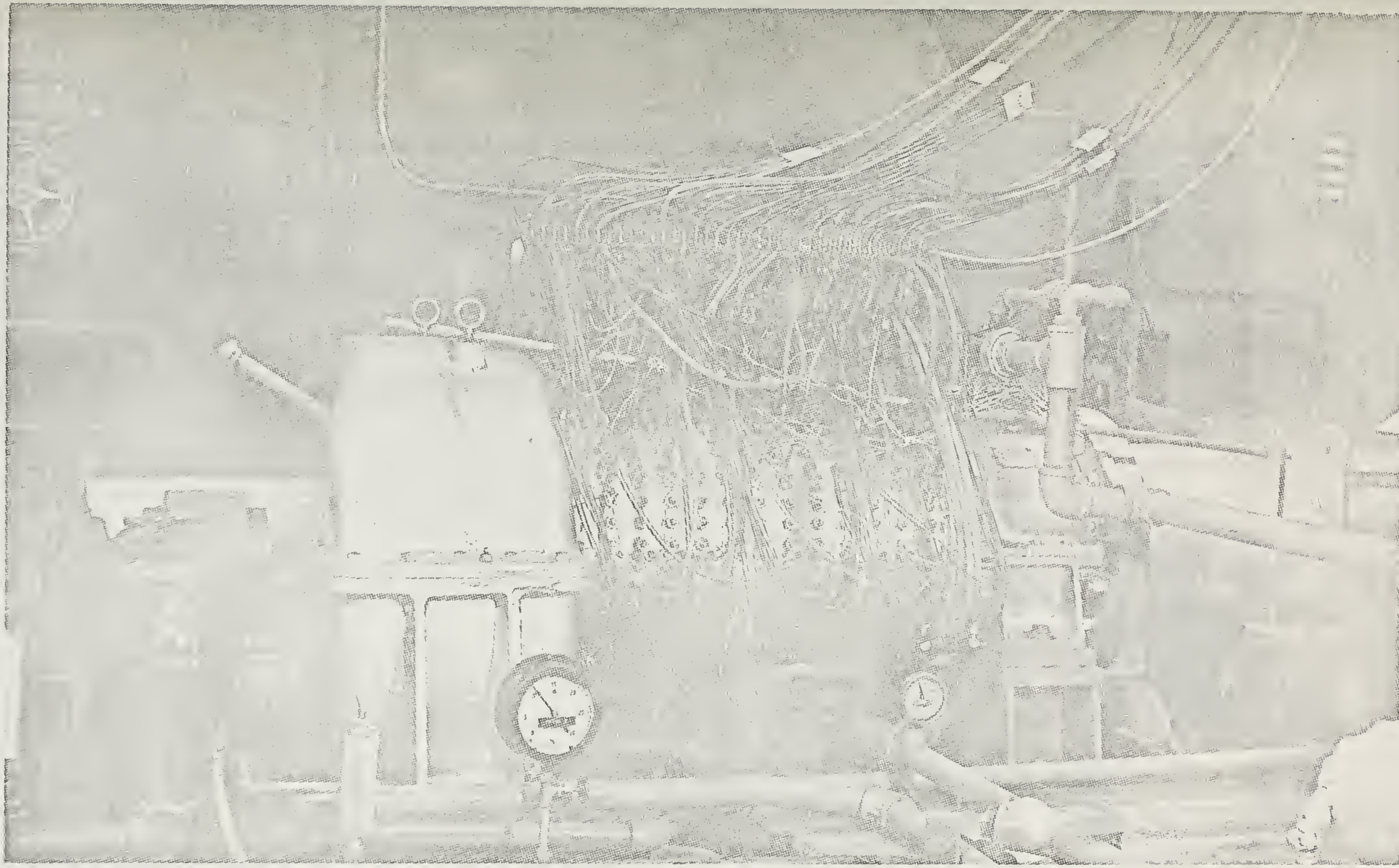

Figure 10: Axial-Flow Compressor

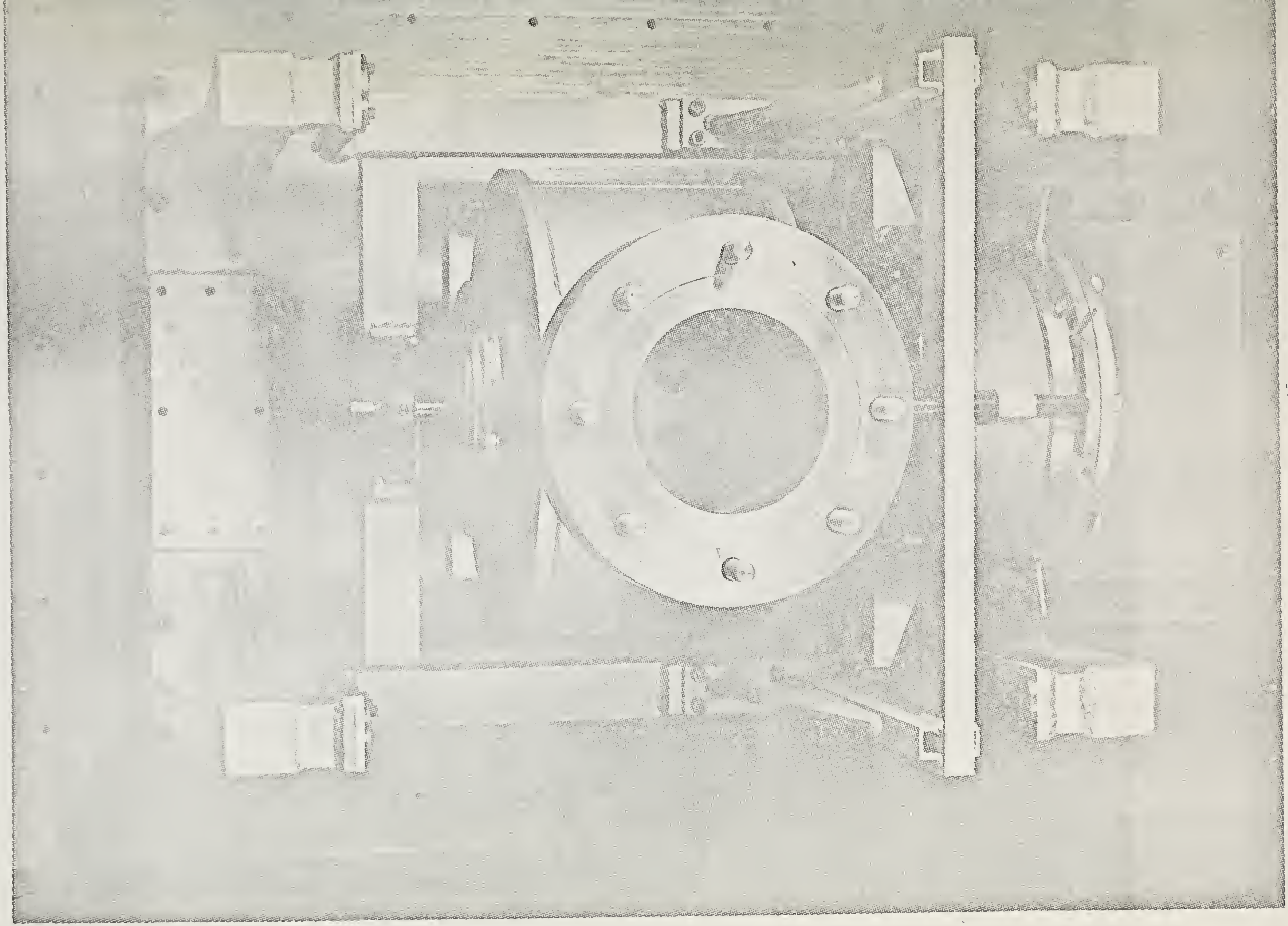

Figure 11: Top View of Transonic Turbine Test Rig.

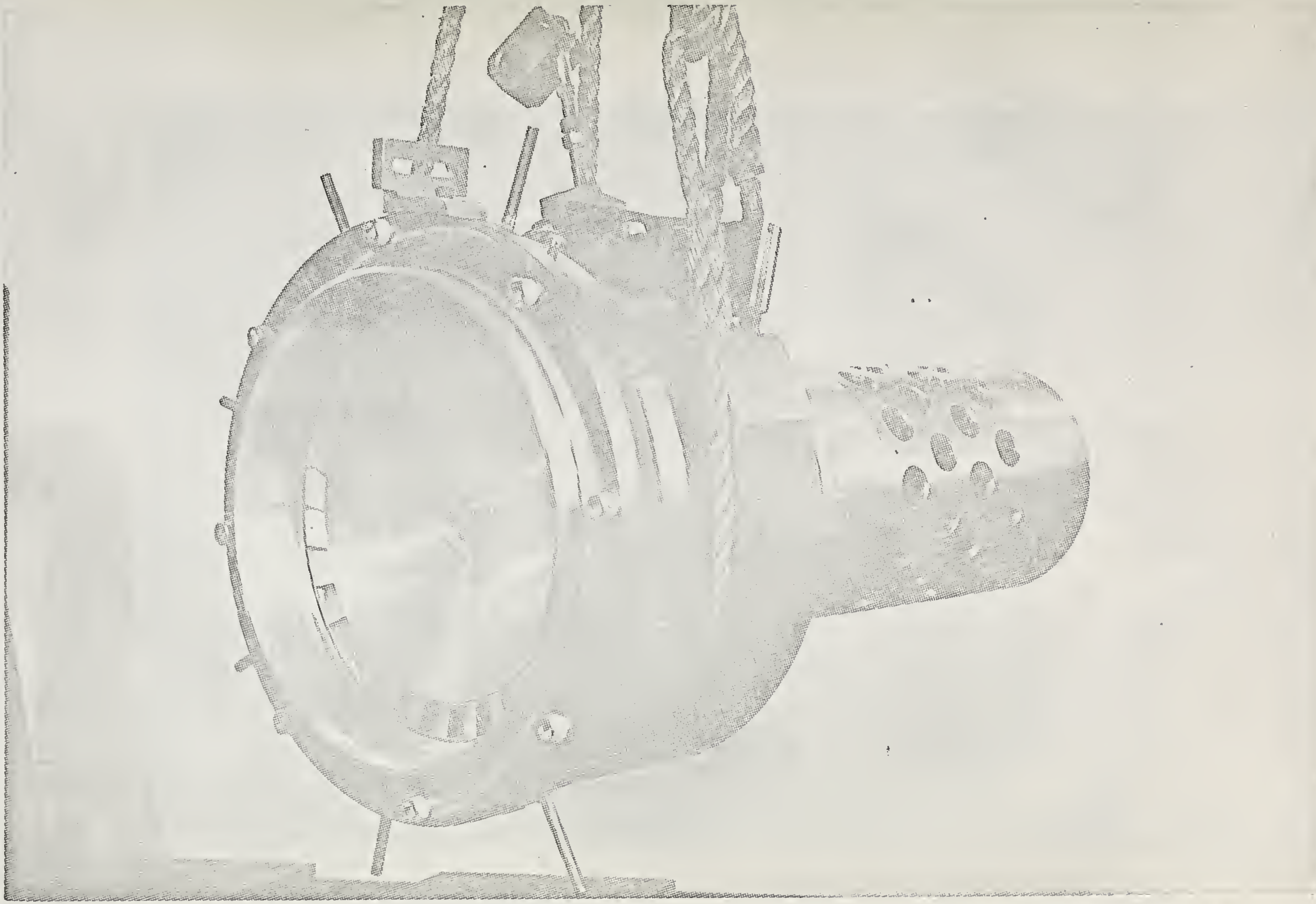

Figure 12: Floating Guide Vane Assembly

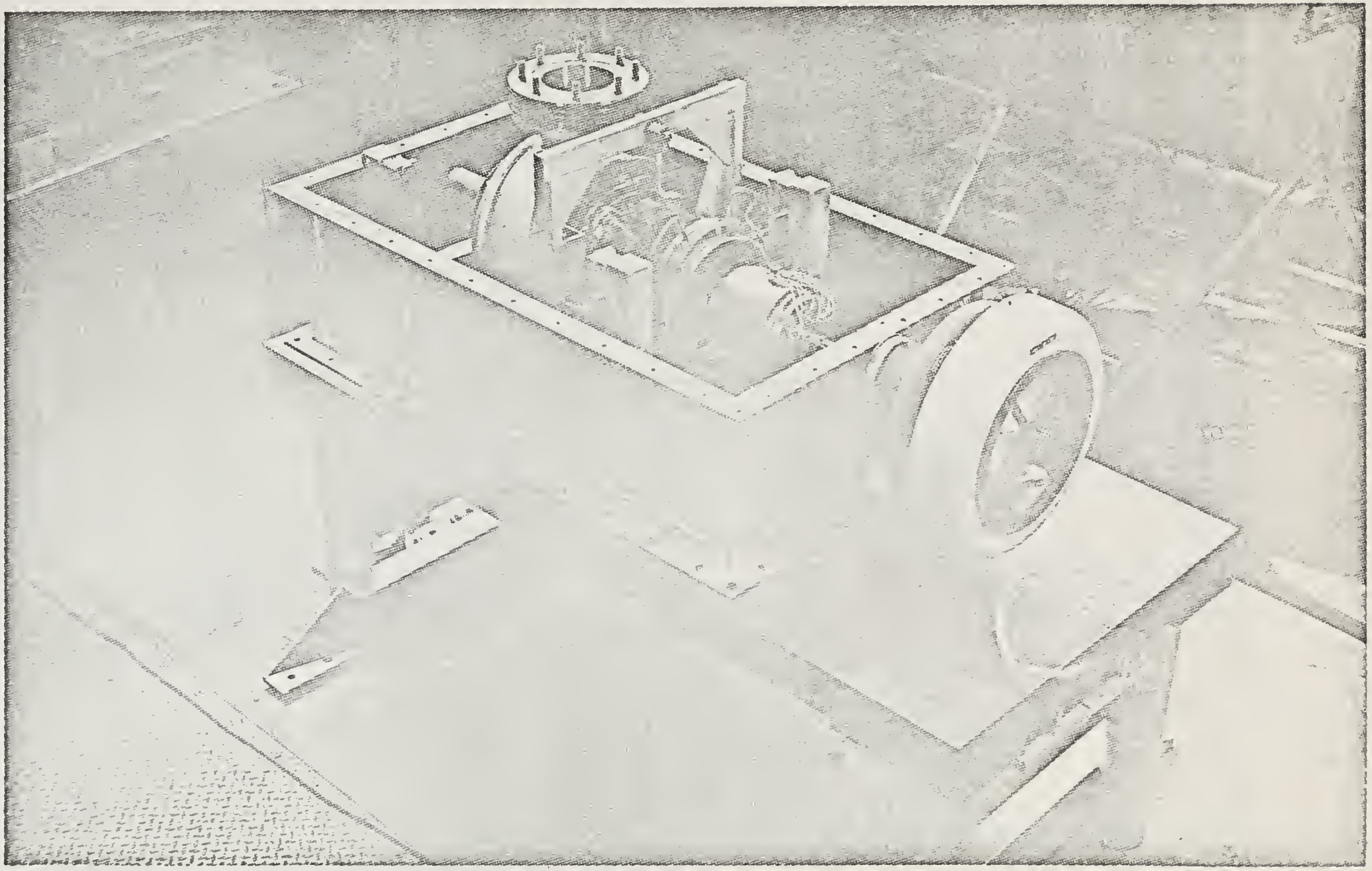

Fig. 13 Assembly of Turbine Test Rig (with tope of hood removed)

Fig. 14 Turbine Test Rig with Exhauster System